



FASTENERS

VITAL IN EVERY INDUSTRY

PUBLISHED BY AMERICAN INSTITUTE OF BOLT • NUT AND RIVET MANUFACTURERS

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*Riveted steel construction means strength
and security on this new highway bridge.*

AMERICAN INSTITUTE OF BOLT, NUT AND RIVET MANUFACTURERS

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INSPECTION—Automatic!

In addition to the usual inspection procedure found in metal working plants, the process of manufacture practised in bolt and rivet plants is an automatic quality inspector of unusual rigidity, covering the raw material used and each part as it progresses to a finished piece.

The raw material must be of uniformly high quality to withstand the upsetting and forming operations. Defects are immediately evident in any piece.

And then a further test exists when the fastener is "brought home" in final assembly.

Fasteners produce a joint automatically inspected from raw material to final assembly, and that is another reason why they are economical and dependable.

Yours truly,

President

QUALITY *control* HELPS MAKE BETTER RIVETS

By HERBERT SCHNEIDER, Metallurgist, Russell, Burdsall & Ward
Bolt and Nut Company



... better and more uniform rivet strength and driving qualities are assured by the closer metallurgical control possible with modern rotary heat treating furnaces ...

SMALL rivets are used in innumerable applications requiring the joining of two or more parts. Riveting is a highly desirable method of permanently joining assemblies because of the low cost, ease and speed of the application, and the certainty of having uniformly sound, strong joints.

Rivets are made with a variety of head shapes, sizes and lengths so that a suitable rivet for the application may be readily available. Rivets may be hot or cold headed, depending on the size and type being produced. The production by cold heading is much greater than by hot heading.

The first step in the production of rivets is to obtain wire of high quality. The rivet manufacturer usually specifies rods of cold heading quality for which a substantial premium is paid. Cold heading quality insures that the most highly developed methods of steel manufacture are employed, which requires the best raw material. Equally important is the selection of the portions of the steel ingots to be used. These ingots are sectioned so that the rods used do not contain solidification imperfections such as pipes and segregation. The former are cracks formed by shrinkage when the last of the molten metal solidifies. There may be segregation of chemical elements which may be determined either by metallographic examination of a section suitably polished and etched, or by chemical analyses. Segregation of chemical elements is a natural result whenever molten metals are allowed to solidify, but

by taking the sections from the ingots, which do not have these defects, harmful segregation and pipes may be prevented from appearing in quality products.

The first step in the manufacture of rivets is to prepare the rods for the drawing operation. Since rods are rolled hot, the surface is covered with a thin layer of iron oxide or scale which acts as an abrasive if not removed. Therefore, the rods are immersed in an inhibited hot sulphuric acid solution of approximately 4 to 8 per cent acid by weight until this scale is removed. They are then water rinsed and dipped in a boiling lime solution, baked and transferred to the wire drawing blocks.

The next step is to draw the wire for cold heading. The rod is reduced about 1/32 in. in diameter by pulling it cold through a die. The physical properties of the rivets are not affected by this drawing since the rivets are subsequently heat treated. In order to obtain suitable header die life and insure close tolerance rivets, it is often necessary to draw the wire to very close tolerances.

The wire drawing die box contains a suitable lubricant to protect the wire drawing die and to impart the best finish to the wire. Special soaps are often used or metal stearates mixed with lime in predetermined percentages. The lime coating on the rods acts as a carrier for the lubricant when the metal is passing through the die.

It is necessary to do a good job on the scale removing operation so that the tool life will not be adversely affected in solid die work. In open die work this is not so important, but insofar as the wire drawing dies are concerned it is always desirable to remove the hot mill scale.

The ultimate tensile strength of the hot rolled rods (maximum load divided by the original cross section in square inches) is in the vicinity of 45,000 psi for A.I.S.I. 1010 steels and will vary somewhat with the chemical components and rolling mill conditions. The physical properties change considerably as the rod is drawn into wire.

Normally the reductions fall between 20 and 30 per cent from rod to wire, and the cold heading operation increases this cold working in the head of the rivet and at the sheared end.

The wire drawing operation serves several useful purposes. It produces a product of uniform cross section within close tolerances and imparts a beneficial coating to the wire. This is very important to solid die life. The metal is also highly stressed which aids in annealing.

A very important step in rivet production is the heat treatment. To produce rivets of the highest quality requires the correct selection of the type of treatment to be given the rivets, proper heat treating equipment, and close supervision.

A brief review of the metallurgical concepts applied to this heat treatment and definitions of several terms are necessary at this point to properly understand rivet heat treatment.

The hot rolled rod is composed of grains or crystals of two metallographic constituents called "ferrite" and "pearlite". These may be revealed to the eye by suitable polishing and etching and observation under a metallurgical microscope. Ferrite is essentially iron with impurities dissolved in it, and therefore the impurities are not seen microscopically. Ferrite appears white. The other constituent, pearlite, is an intimate mechanical mixture of ferrite and iron carbide called cementite. This constituent appears dark. When steel is cooled at a suitable cooling rate from certain temperatures, the grains are said to be equiaxed, that is, they have equal diameters in all directions. When the finishing temperatures in the rod mills (temperature at the last roll stand) are high enough, the cooling rate is such that the structure of the hot rolled rod appears equiaxed.

When the rods are pulled through the wire drawing die, the grains are gradually distorted and as the degree of reduction is increased, the grains increase in length in the direction of the pull on the wire, and decrease in cross section. This is referred to as cold working and is accompanied by the change of physical properties tabulated in Table I.

TABLE I
EFFECT OF COLD DRAWING
ON PHYSICAL PROPERTIES

Diameter, in.	Reduction of area from rod, %	RESULT OF TENSILE TESTS		
		Tensile strength, psi	Elongation in 2 in., %	Reduction in area, %
0.361	0	47,000	45.5	76
0.340	9.0	54,250	29	72.5
0.321	20.8	61,500	14	69
0.301	29.7	67,000	12.5	67
0.279	39.6	75,000	12	61.5
0.256	49.5	77,200	6	57.5

TABLE II
EFFECT OF ANNEALING TIME
ON PHYSICAL PROPERTIES

1300° F. SALT BATH

Diameter, in.	Annealing time in salt bath, minutes	RESULTS OF TENSILE TESTS		
		Tensile strength, psi	Elongation in 2 in., %	Reduction in area, %
0.256	2	48,800	41	71.8
0.256	4	48,200	43	75
0.256	8	47,500	45	78
0.256	16	48,300	44	75
0.279	2	48,800	42	71.5
0.279	4	47,650	42	73.5
0.279	8	46,650	42	73.5
0.279	16	47,450	39	72.5
0.301	2	49,500	36	71.7
0.301	4	48,000	39	76.7
0.301	8	46,450	42	79.2
0.301	16	47,500	47	78
0.321	2	58,000	30	61.2
0.321	4	46,680	37	73
0.321	8	45,700	43	76.5
0.321	16	46,750	48	78.8
0.340	2	59,400	29	64.5
0.340	4	50,800	30	72.5
0.340	8	45,800	46	74.8
0.340	16	45,100	47	77.6
0.361	2	49,400	40	64.8
0.361	4	50,700	36	74.7
0.361	8	49,700	44	78.8
0.361	16	48,600	43	67.3

The stresses produced in cold heading rivets are superimposed on the stresses produced in the wire drawing operation. The heat treatment of rivets is designed to eliminate these cold working stresses and reduce the hardness so that they will possess the proper toughness and ease of upsetting. The temperature at which the elongated grain becomes equiaxed is called the "recrystallization" temperature. The recrystallization temperature is controlled by the amount of cold working prior to the heat treatment. The greater the cold work, the lower the recrystallization temperature. The recrystallization of the ferrite crystal is quite rapid at 1300 F and above, but the pearlite remains elongated until higher temperatures are reached.

The pearlite grains are equiaxed by a different mechanism than the ferrite. When on heating, the

steel reaches a temperature called the lower critical point (about 1340 F) the pearlite transforms into another constituent called austenite. As the temperature is further raised, the austenite expands and dissolves the ferrite until the entire structure is austenite. This is around 1640 F for A.I.S.I. 1010 steel. The rate at which the steel cools to room temperature determines how the new pearlite and ferrite grains are formed, their size, and distribution, and the resulting physical properties.

It is apparent by now that cold worked, low carbon steel may be heat treated so that only the ferrite is equiaxed, called slack annealing, or a slightly higher temperature may be used which recrystallizes the ferrite on heating and forms new pearlite grains on cooling, or the critical range may be exceeded so that new ferrite and pearlite grains are formed as the steel cools through the critical range. The last procedure is called full annealing, since by this method entirely new ferrite and pearlite grains are formed. Air cooling from above the upper critical temperature is called "normalizing" while slow cooling is referred to as "full annealing".

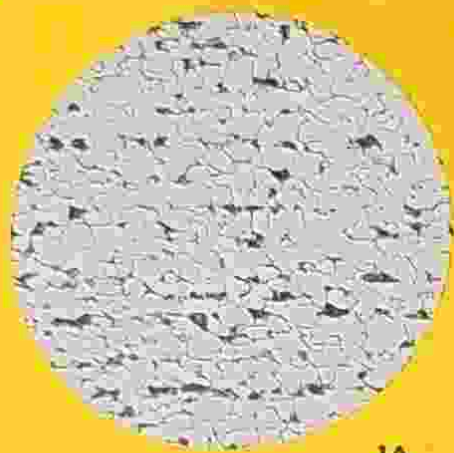
To determine the time required to recrystallize at various temperatures, the following experimental procedure was carried out. A 23/64 in. diameter A.I.S.I. 1010 rod (nominally 0.359 in. diameter) was drawn to 0.340, 0.321, 0.301, 0.279 and 0.256 in. diameters and samples of each cut. This represents approximately 10, 20, 30, 40 and 50 per cent reduction from rod to wire. Samples of each wire were annealed in salt baths, the bath temperature being 1300 F and 1670 F in time intervals as short as 30 seconds and as long as 16 minutes, followed by cooling in still air. The results of the physical property checks are shown in Tables II and III.

These tables indicate that for some reductions as little time as 1/2 minute is sufficient to eliminate the effects of cold work. Note the physical properties of the 0.321 in. wire for 1/2 minute at the high temperature. Even two minutes at 1300 F show fairly complete elimination of cold work. This refers to total time in the salt bath so that the wire probably was not actually at 1300 F for more than 1/2 minute. Figure 1 shows the effect of cold work on the grains. Figure 2 compares the recrystallization structure after annealing from two temperature levels and compares final structure with that of rivet subsequent to wire drawing and rivet heading operations.

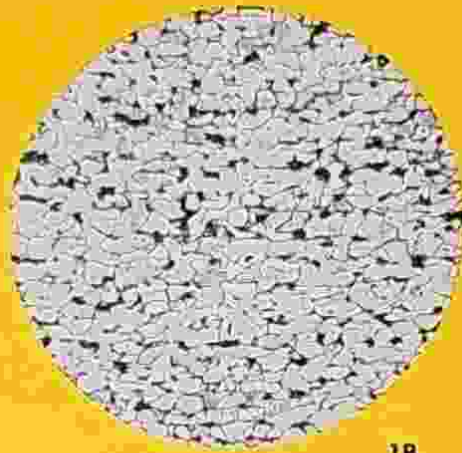
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TABLE III
EFFECT OF ANNEALING TIME
ON PHYSICAL PROPERTIES
1640° - 1670° F. SALT BATH

Diameter, in.	Annealing time in salt bath, minutes	RESULTS OF TENSILE TESTS		
		Tensile strength, psi	Elongation in 2 in., %	Reduction in area, %
0.256	1/2	51,000	41	71.8
0.256	1	50,400	44	72.6
0.256	2	50,400	40	70.6
0.256	4	50,600	40	72.2
0.279	1/2	52,400	30	71.8
0.279	1	50,400	38	70.1
0.279	2	50,100	41	72.7
0.279	4	50,400	42	72.2
0.301	1/2	49,500	44	73.1
0.301	1	49,800	46	74.8
0.301	2	49,500	43	72.2
0.301	4	49,650	41	73.1
0.321	1/2	48,650	49	75.6
0.321	1	48,900	46	73.4
0.321	2	48,800	43	73.7
0.321	4	49,450	42	73.7
0.340	1/2	51,200	36	70.0
0.340	1	49,700	40	73.2
0.340	2	49,400	42	72.6
0.340	4	48,700	43	74.8
0.361	1/2	50,900	40	76.2
0.361	1	50,600	40	73.2
0.361	2	49,700	45	74.4
0.361	4	49,900	44	75.7



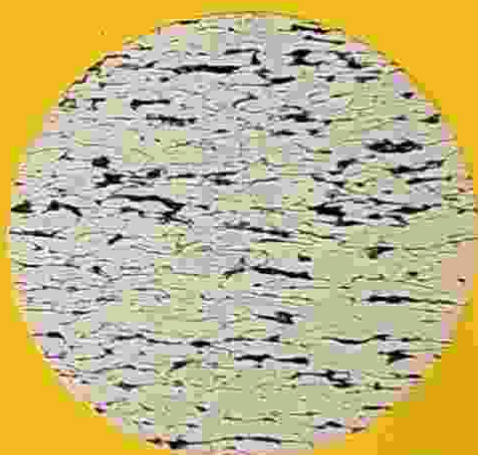
1A



1B



1C



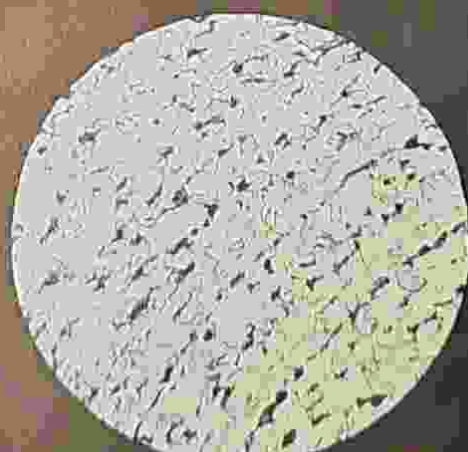
1D

Figure 1—No effect of cold work is visible (upper left), when wire is drawn 9 per cent. (Upper center) a little effect of cold working can be seen when the wire is drawn 20 per cent. Elongation of ferrite and pearlite can be seen when the wire is drawn 30 per cent (upper right). On the center right, the marked elongation of ferrite and pearlite grains due to cold working can be readily seen when the wire has been drawn 50 per cent. Magnification 100X.

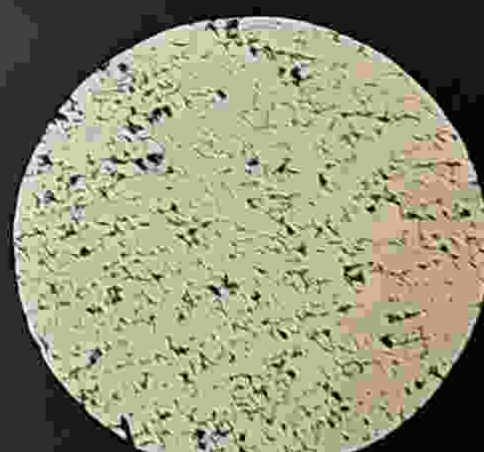
Figure 2—The cold working has a very marked effect on the rivet as shown by the micrograph in the lower left diagram. The lower right diagram shows the effect of normalizing the rivet, by annealing in the salt bath at 1640 F for 2 minutes, followed by air cooling. Both ferrite and pearlite are recrystallized, and there is a more random distribution of the finer pearlite grains than in the lower center diagram which shows the rivet structure after annealing in the salt bath at 1300 F for two minutes and air cooled. In the latter case, although the ferrite has recrystallized, the pearlite remains elongated (slack annealed).



2A



2B



2C

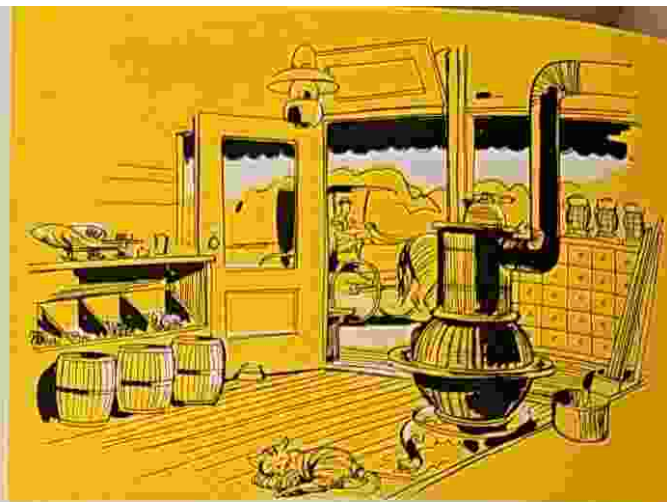
Bolts at the Four Corners

by J. H. MIZE

Secretary and Sales Manager

Blish, Mize and Silliman Hardware Company

Atchison, Kansas



... the article "Bolts to the Four Corners" in last month's issue of FASTENERS analyzed the function of the distributor from the manufacturer's viewpoint ... the following article continues this analysis from the distributor's viewpoint ...

BOLTS, nuts, and screws have long been synonymous with hardware. No item in the hardware field has been so consistently important as bolts, and no item, probably, is so often written on mail and salesmen's orders in the hardware retail store. Dealers in hardware are definitely fastener conscious. All sizes are in large demand, with some smaller diameters always on the "want book." Hardly an order passes through a jobbing warehouse that does not include substantial quantities of fasteners.

In 1902 our general catalog listed 76 stock sizes of machine bolts, and today we list several hundred, all of which are active. Twenty years ago no retailers and few jobbers carried machine and carriage bolts in diameters larger than $\frac{1}{2}$ in. Now most good dealers carry stocks of carriage bolts up to $\frac{5}{8}$ and $\frac{3}{4}$ in. in diameter and machine bolts up to and including 1 in. Twelve and fourteen inches used to be the extreme stock lengths, but of late years it is common practice for retailers to stock lengths up to 24 inches. All this change in stocking of sizes has come about because of the increasing use of heavy farm machinery, such as tractors and other large farming equipment, plus road maintenance machinery, oil field equipment, etc. Likewise, the unusually heavy demand for small bolts within the last few years is largely due to the construction of truck bodies by the farmer, rancher, produce hauler, and home trailer maker. It is not unusual to see a consumer walk into a retail store and buy $\frac{1}{4} \times 3$ in. carriage bolts in lots of 300 or more.

The popularity of the cap screw and semi-finished hex nut, particularly in the agricultural sections, is substantially on the increase. Some of our retail customers have actually expressed the opinion that

the trend is to "steel bolts" and away from machine bolts. One southern salesman reports that he has at least three hardware stores that are seriously thinking of discontinuing machine bolts in favor of cap screws. Much of the new farm machinery uses steel bolts and the farmer is being educated to use them rather than machine bolts. The $\frac{9}{16}$ in. cap screw, in particular, is fast becoming a popular diameter as several farm machinery companies use it on their equipment and this particular size is not readily available in the standard machine bolts.

The aggressive bolt distributor attempts to stock a complete range of sizes and types of fasteners. In some cases this effort to serve the retailer by having variety is not as economical as it might be if only turnover items were stocked. The distributor, however, is a service institution and it pays in the long run to take care of the occasional unusual request. Standardization on the part of manufacturers within recent years has materially helped in weeding out burdensome inventories of special sizes.

The war and post-war period made retail dealers extremely conscious of bolts with attached nuts. They now dislike bolts with separate nuts and if necessary are willing to pay extra for the attached nuts. It is a real convenience and time saver for the dealer to get his bolts with nuts attached.

The retail hardware dealer likes to sell bolts and rivets and nuts and screws for several very fundamental reasons. One reason, of course, is the profit involved, for fastener items have always carried a satisfactory profit margin. Seldom is the consumer price conscious on bolts, whereas rope, on the other hand, at fifty cents per pound on today's retail market brings decided sales resistance. Bolts selling either by the pound or by the bolt list require little sales effort even though some 75 per cent higher than in many pre-war years. An Oklahoma retail hardware dealer recently told the writer that his bolt stock was the most important stock in his store from a turnover and profit angle—and this

man has made a small fortune in the hardware business.

A well rounded stock of bolts is a trade builder as it increases store traffic. Traffic generating therefrom gives the dealer an opportunity to sell related hardware items such as wagon box hardware, machinery repairs, belting and other "farm" items. Another reason for the popularity of fasteners in the eyes of the retailer is the fact that they are easily stocked and take up a minimum of space—in fact in many stores they have become a "self service" item. Furthermore, a well rounded retail stock of bolts reduces seasonal variance in sales; for a good assortment of fasteners will sell throughout the year. Increasing agricultural and industrial uses in all sections of the country have brought this about. Finally, bolts and nuts and most other fasteners are one of the few items that have stayed with the hardware retailer. They have not drifted to the drug and department store; nor have the chains apparently affected the sales at retail.

The wholesale hardware salesman or "bolt and nut peddler" (as hardware salesmen are often called) likes to think that he controls the bolt and nut business in his territory. Every salesman likes to sell bolts, and one Colorado man said once that bolts and screws were his "binding" friends and a tie that closely linked the retailer and his distributor. The volume on fasteners of all types is so large that this material is an order starter for salesmen; it pro-

vides shipping weight for small orders, and adds volume to the salesman's sales.

The wholesaler appreciates his fastener volume. Proper selling discounts are important, and all jobbers like to feel that their discounts to the trade are well in line with prevailing competition. The common practice is to sell by list and discount, although many distributors feel that billing net to the retailer is more convenient for the customer. Proper stock methods and proper binning are most important, and distributors are constantly on the alert for better and newer methods of getting bolts to their packing rooms for quicker and better service to dealer customers. The distributor normally purchases his fastener requirements from relatively few sources; for splitting purchases among too many manufacturers reduces his importance as a customer.

This article has dwelt primarily on the retail aspect of fasteners and their importance to distribution. However, the basic importance of the line to hardware distributors is only as important as it is to his customers. If the line is well thought of by the retailer, the importance is felt by the jobber, and the line of bolts would not be important to a distributor if the retailer failed to promote it. Needless to say, all hardware distributors consider a complete line of fasteners a "must"; and the line ranks at the top among important jobber promotions.



Figure 1—Indicative of the importance that bolts play in the operations of a hardware dealer is this view of a corner of the storage room for bolts maintained by the author's company.

Development of ASA Standard for Slotted and Recessed Screw Heads B 18.6-1947

by F. P. TISCH
Chief Engineer
Pheoli Manufacturing Company

... This report by the chairman of Subcommittee No. 3 of the sectional committee on the Standardization of Bolt, Nut and Rivet Proportions illustrates how much tedious labor and time are required to develop a new standard and obtain its approval ...

INTERCHANGEABILITY, the foundation on which modern mass production is based is a result of standardization. Standardization in turn is founded on the agreement of qualified individuals as to what constitutes the most practical and economical specifications covering an extensively used product. The evolution of nationally accepted standards demonstrates the necessity of definite specifications for an article, as the existing confusion due to variation in designs, sizes, and qualities becomes apparent.

When such a condition exists in an industry, manufacturer's organizations or engineering societies associated with the product usually institute a standardization program. Since a standard approved by the American Standards Association is more widely accepted, these organizations act as sponsors for the development of the standard and form a committee, members of which are chosen because of their knowledge and experience in the manufacture and use of the product. The usual detailed procedure in formulating and obtaining approval for a typical standard is illustrated in this article.

The period of increased mechanical activity before World War II made it more and more evident that the former American Standard, covering slotted head proportions for machine, cap, and wood screws, was inadequate for the newer uses of threaded products. Recognition of this situation prompted the American Society of Mechanical Engineers and the Society of Automotive Engineers to create a committee to consider the existing standards and develop such additional standards as were necessary.

This committee was selected from men in the

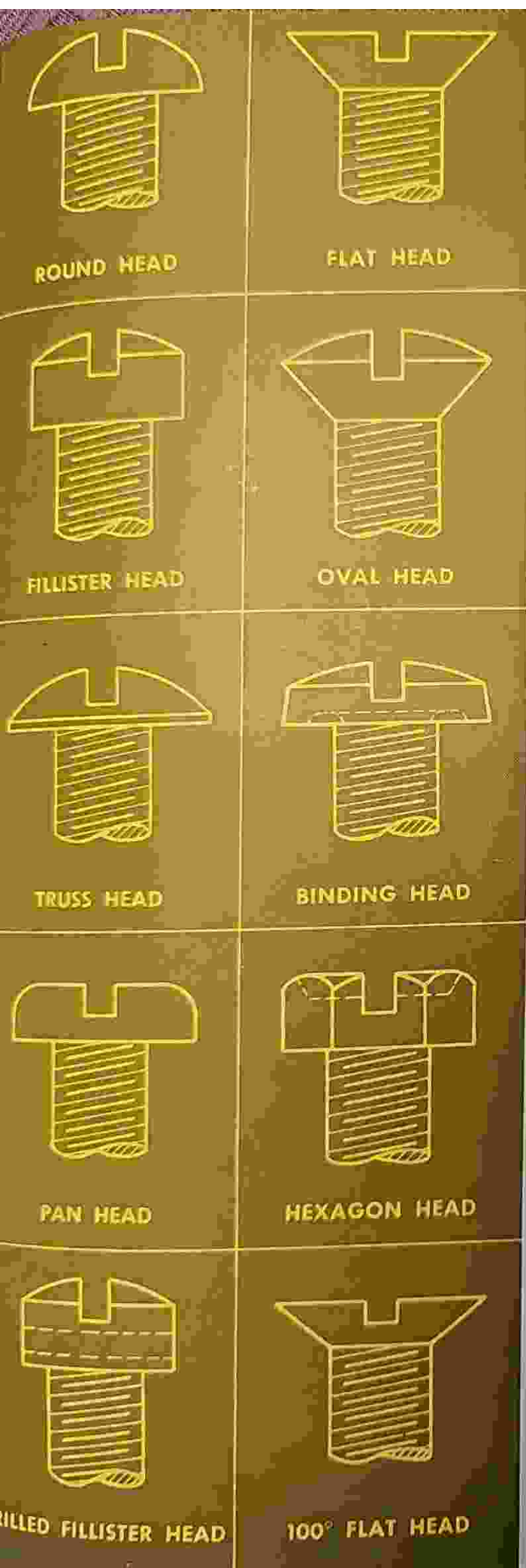
automotive, aircraft, electrical, railway, and stove industries, as well as, manufacturers of the screw products. This committee, known as subcommittee No. 3 of ASA B18, held its organization meeting on the 14th of April 1942. The scope of the subcommittee was defined as the complete product specification of all screws that were commonly driven by suitable slots or recesses. Included in these products were machine, cap, wood, tapping and headless set screws.

In order to expedite the work, two subgroups were appointed; one to cover all of the head, slot, and recess dimensions, and the other group to cover all of the other elements of the various products. Ten meetings were held during the ensuing five years, resulting in the tentative standards covering machine, cap, wood, tapping, and slotted headless set screws.

A tentative proposal for the revision of this standard was distributed to industry for criticism and comment in April 1945. It was also distributed to the members of Subcommittee No. 3 in advance of this committee's meeting on June 8th. At succeeding meetings of a special editing committee further revisions were incorporated resulting in a May 1946 draft which was prepared for use in securing the letter ballot vote of the Sectional Committee. The Screw Industry Standards Committee, U. S. Machine Screw Service Bureau, cooperated in the preparation of the data and approved the work developed at its meeting on November 16, 1945.

Following the approval of the Sectional Committee, the proposal was submitted to the sponsor organizations for their approval. Formal acceptance of these bodies followed and the American Standards Association gave its approval and designation as an American Standard on April 22, 1947.

This standard differs from the one previously published in April 1930, in that the range of sizes for machine screws is from size 0 to 34 in. diameter,



and several new head styles have been added, as well as dimensions for recessed heads, tapping screws, and slotted headless set screws.

The inclusion of this additional data is not intended to imply that all of the products described in the standard are stock production sizes. Consumer interests are requested to consult manufacturers' catalogs for lists of stock production sizes for the various types of products.

In order to reduce the necessity of several screw drivers, for each screw, it was found advisable to specify a single slot with adequate tolerances for each size or diameter of screw regardless of type or head style.

General data were established covering such features as angularity and eccentricity of heads, fillets, slot depth measurement, unthreaded shank diameter, length tolerance and standard method of measurement, thread lengths, point specification, classification of threads, and finish.

Round, flat, fillister, and oval head machine screws were included in the former standard. Modern requirements for machine screws justified the addition of truss, binding, pan, hexagon, drilled fillister, and 100 degree flat head types. The wide usage of cross recessed heads led to the inclusion of the two types in the standard.

The flat head cap screw table is extended from the $\frac{3}{4}$ in. size to include the $\frac{7}{8}$ and 1 in. sizes, and the former designation of button head was changed to round head. Dimensions for the unthreaded shank of cap screws were added to all tables.

The tables for wood Screw dimensions were improved by the addition of the number of threads per inch for each size.

Complete specifications for three types of tapping screws have been established, namely; type A, gimlet point; type B, blunt point; and type C which has a machine screw thread with a blunt tapered point. The head styles for tapping screws are in-

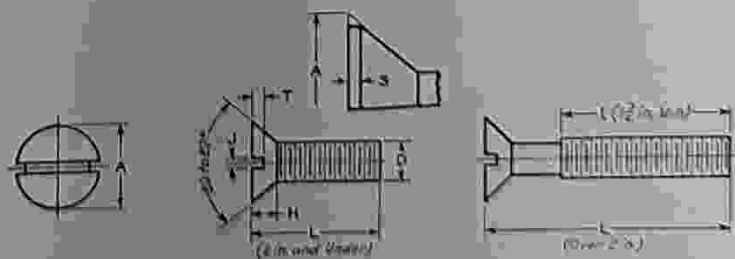


Figure 1—Dimensions standardized for slotted flat head machine screws.

TENTATIVE SPECIFICATIONS FOR STEEL MACHINE BOLTS AND NUTS AND TAP BOLTS A.S.T.M. DESIGNATION: A307-47T

SCOPE

1. These specifications cover machine bolts and nuts, and tap bolts for general application.

PROCESS

2. (a) Steel for the manufacture of bolts shall be made by either the open-hearth or electric-furnace process.

(b) Steel for the manufacture of nuts shall be made by the open-hearth, electric-furnace or acid-bessemer process.

FABRICATION

3. (a) Bolts may be produced by hot or cold forging of the heads or machining from bar stock. Cold-forged bolts shall be subsequently heated to a temperature sufficient to meet the head test specified in Section 8.

(b) Threads shall be rolled or cut, or cold formed.

(c) Nuts shall be produced by hot pressing, cold punching, cold forging or machining from bar stock.

CHEMICAL COMPOSITION

4. (a) Steel for the manufacture of bolts and nuts shall conform to the following chemical composition:

	Bolts	Nuts
Phosphorus, maximum, per cent	0.06	0.13
Sulphur, maximum, per cent	0.15	0.23

(b) Resulphurized material is not subject to rejection based on check analysis for sulphur unless misapplication of steel is clearly indicated.

(c) Bolts and nuts are customarily furnished from stock, in which case individual melts of steel cannot be identified.

PHYSICAL PROPERTIES

5. (a) Where tested in full size for tensile properties, bolts shall meet the total breaking loads specified in Table I.

(b) When bolts are too large for full size tension testing, turned specimens shall have a minimum tensile strength of 50,000 psi.

(c) When bolts are too short for tension test, they shall show a minimum hardness of 92 Brinell (58 Rockwell B).

(d) Nuts are not subject to mechanical tests. However, all standard nuts except Jam nuts will meet a proof strip load as shown in Table I.

TENSION TEST

6. (a) Tension tests of bolts shall be taken preferably on the finished bolt with the load applied between the

head and nut or suitable fixture, either of which shall have sufficient thread engagement to develop the full strength of the bolt. The nut or fixture shall be assembled on the bolt leaving at least three bolt threads exposed within the grip. Fig. 1 illustrates the method for testing a full size specimen. If failure occurs by threads stripping before reaching the minimum required tensile load the test shall be discarded.

(b) Where the bolt or thread length is too short for a tension test, acceptance shall be determined by the hardness test described in Section 7.

(c) The tensile strength determination shall be based upon the cross-sectional area of the bolt where the break occurs. The cross-sectional area of the threaded portion used in the determination shall be as shown in Table I.

(d) Where equipment of sufficient capacity is not available for testing bolts in full size as described in Paragraph (a), tests may be made as follows: Bolts $\frac{7}{8}$ to $1\frac{3}{8}$ in. in diameter, inclusive, shall have their shanks turned concentric with the axis of the bolt as shown in Fig. 2, leaving the head and the threaded section of the bolt intact. The load shall be applied between a nut or suitable fixture and bolt head. For bolts $1\frac{1}{2}$ in. and over in diameter, a standard test specimen shall be turned from the bolt, having an axis midway between the center and the outside surface of the bolt shank as shown in Fig. 3, with a diameter and gage length as shown in Fig. 4.

HARDNESS TEST

7. (a) Hardness of bolts shall be measured on a transverse section through the threaded portion of the bolt at a point one-eighth of the nominal diameter from the axis of the bolt.

(b) Hardness measurements shall be made preferably by the Brinell method. Where sections or hardness range do not permit the use of Brinell testing, the Rockwell method may be used. The preparation of the sample and the method of performing the test shall conform to the Standard Method of Test for Brinell Hardness of Metallic Materials (A.S.T.M. Designation: E 10) or the Standard Methods of Test for Rockwell Hardness and Rockwell Superficial Hardness of Metallic Materials (A.S.T.M. Designation: E 18). Conversion of values between these systems shall conform to the Standard Hardness Conversion Tables for Steel (A.S.T.M. Designation: E 48). Conversion of hardness measurements to tensile strength shall conform to the conversion table published by the S.A.E.

HEAD TEST

8. During the tension test specified in Section 6 (a) failure shall occur in the shank or the threaded section and not at the juncture of the head and shank.

STRIPPING TEST

9. (a) The sample nut shall be assembled on a hardened, threaded mandrel and the load specified in Table I applied to the nut. Threads of the nut shall not strip at this load. If the threads of the mandrel are damaged during the test, the test shall be discarded.

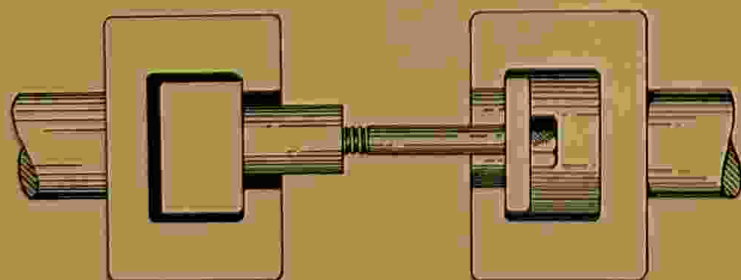


Figure 1—Tests on full size specimens shall be made in a rig as illustrated by this sketch.

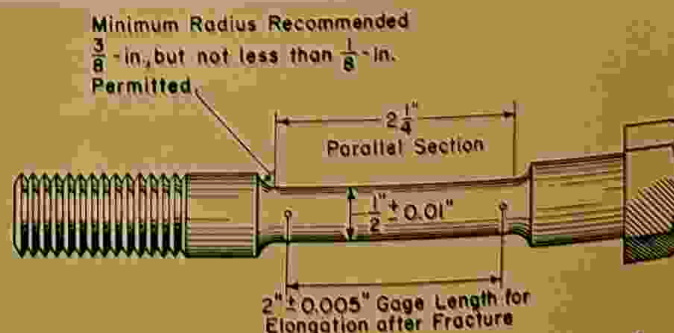


Figure 2—Large specimens may be turned down as indicated in order to utilize available testing machines.

(b) The mandrel shall be threaded to the American Standard Class 3 tolerance except that the major diameter shall be basic with a minus tolerance of 0.002 in.

(c) If the unit tensile stress developed in the mandrel is required, the loads thus obtained shall be divided by the mean thread area as given in Table I.

NUMBER OF TESTS

10. (a) The requirements of these specifications are those met in continuous mass production for stock during which the manufacturer has made such sample inspections as to insure normally that the material is controlled within the specified limits. For this reason additional tests by the manufacturer of individual shipments of material are not contemplated.

(b) One tension test may be made from each lot of bolts. Unless otherwise agreed upon, a lot shall consist of 5000, or fraction thereof, for diameters of 1 in. and smaller, and 2,500, or fraction thereof, for diameters larger than 1 in.

(c) If any test specimen shows defective machining it may be discarded and another specimen substituted.

(d) The stripping test and hardness test may be made on nuts selected as follows: Two nuts per keg for sizes $\frac{1}{2}$ in. and smaller; one nut per keg for sizes larger than $\frac{1}{2}$ in. up to and including $1\frac{1}{2}$ in.; one nut per two kegs for sizes larger than $1\frac{1}{2}$ in.

RETESTS

11. Should any sample from the same lot fail to meet the requirements of a specified test, double the number of samples shall be tested, in which case all of the additional samples shall meet the specifications.

DIMENSIONS

12. (a) Threads shall be the coarse thread series as specified in the American Standard for Screw Threads (A.S.A. No.: B1-1-1935), having a Class 2 tolerance for bolts and a Class 1 tolerance for nuts $\frac{1}{4}$ to $\frac{3}{8}$ in. in diameter, inclusive, and a Class 2 tolerance for nuts $\frac{3}{4}$ in. and larger in diameter.

(b) Bolt heads and nuts shall be Unfinished American Standard Regular as specified in American Standard for Wrench Head Bolts and Nuts (A.S.A. No.: B 18-2-1941).

MARKING

13. Bolt heads shall be marked (by raised or depressed mark at option of the manufacturer) to identify the manufacturer. The manufacturer may use additional markings for his own use.

INSPECTION

14. (a) If the inspection described in Paragraph (b) is required by the purchaser it shall be specified in the inquiry, order or contract.

TABLE I—PHYSICAL PROPERTIES

Bolt size and thread per inch	Mean thread area, sq. in. (2)	Breaking load of bolts, min. lb. (3)	Stripping load of nuts, min. lb. (4)
$\frac{1}{4}$ —20	0.0317	1,600	2,900
$3/16$ —18	0.0522	2,600	4,700
$1/2$ —16	0.0772	3,900	6,900
$7/16$ —14	0.1060	5,300	9,500
$1/2$ —13	0.1416	7,100	12,700
$9/16$ —12	0.1816	9,100	16,300
$5/8$ —11	0.2256	11,300	20,300
$3/4$ —10	0.3339	16,700	30,100
$7/8$ —9	0.4611	23,100	41,500
1 — 8	0.6051	30,300	54,500
$1\frac{1}{8}$ —7	0.7627	38,100	68,600
$1\frac{1}{4}$ —7	0.9686	48,400	87,200
$1\frac{3}{8}$ —6	1.1539	57,700	104,000
$1\frac{1}{2}$ —6	1.4041	70,200	126,500
$1\frac{3}{4}$ —5	1.8982	94,900	171,000
2 — 4 $\frac{1}{2}$	2.4971	125,000	224,500
$2\frac{1}{4}$ —4 $\frac{1}{2}$	3.2464	162,500	292,000
$2\frac{1}{2}$ —4	3.9977	200,000	360,000
$2\frac{3}{4}$ —4	4.9237	246,000	443,000
3 — 4	5.9660	298,500	537,000
$3\frac{1}{4}$ —4	7.0973	355,000	639,000
$3\frac{1}{2}$ —4	8.3269	416,500	749,000
$3\frac{3}{4}$ —4	9.6547	482,500	869,000
4 — 4	11.0806	554,000	997,500

(2) Area at mean of mean root and pitch diameter of Class 3 external thread.

(3) Based on a tensile strength of 50,000 psi. min.

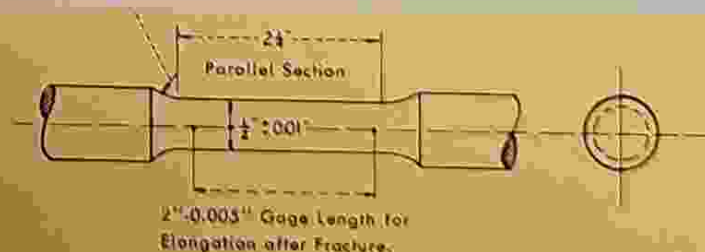
(4) Based on a tensile strength of 90,000 psi. min.

(b) The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works that concern the manufacture of the material ordered. The manufacturer shall afford the inspector all reasonable facilities, without charge, to satisfy him that the material is being furnished in accordance with these specifications. All tests (except check analysis) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

Minimum Radius Recommended

$\frac{3}{8}$ in., but not less than $\frac{1}{8}$ in.

Permitted



Note: The gage length, parallel section and fillets shall be as shown, but the ends may be of any shape to fit the holders of the testing machine in such a way that the load shall be axial.

Figure 4—Test specimens will conform to the dimensions outlined in this figure.

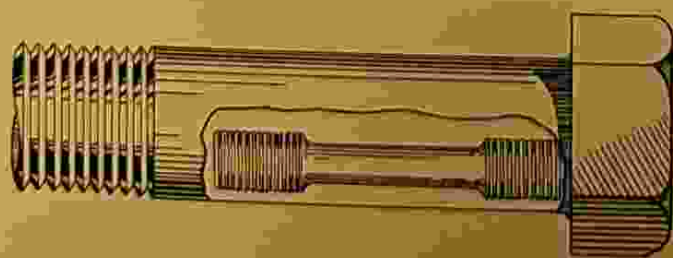
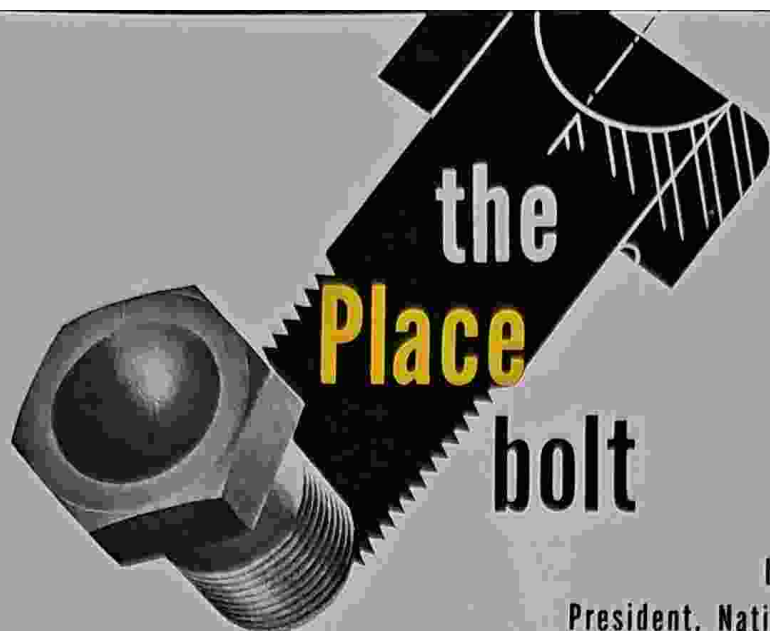


Figure 3—For bolts $1\frac{1}{2}$ in. diameter and over, test specimens will be taken from the section of the bolt indicated in the drawing.



by C. L. BRACKETT
President, National Machine Products Company

... in reviewing fasteners of the bolt or cap screw type, the head construction here described is perhaps unique in the fact that in it is a controlled spring element which is adaptable to either type fastener ...

"WAISTED" screws have been in use for a considerable length of time. In this type screw, the greater part of the shank or unthreaded portion is turned down to the bottom diameter of the thread, and because of the elastic yield, the factor of safety against loss of initial tension is increased. This loss probably results from small plastic deformation of the thread faces under load. The Place type head similarly provides an elastic yield without the reduction of shank diameter.

In construction, this type head is formed with a centrally disposed recess in its upper face, and a circular recess adjacent to the shank in its underface. Between the radii of these recesses a diaphragm of metal is disposed (see checking dimension X in Figure 1) in the form of the conical surface of a frustum. The area of this conical surface is so proportioned with respect to the thread root area that the latter remains, as in the case of a conventional bolt, the weakest section.

When under load, the uninterrupted wall around the recess at the top of the head offers full resistance to an inward movement, and in consequence permits the diaphragm of metal to function as a controlled spring element.

The spring element, when the bolt is tightened, performs two functions. Through the added elasticity, the resistance between the bearing surfaces against loosening is maintained over a considerably greater arc, thus providing the fastener with the desired "locking feature". Furthermore, the more favorable stress distribution in the volume of the head insures the bolt to a very noticeable degree against shock or impact failure.

Prior to the now large production usage of these bolts, several interesting tests were carried out, one of which may be of particular interest to the reader. In June of 1941, a two dimensional photoelastic analysis, comparing Place type heads with conventional bolt heads, was undertaken at the Stevens Institute of Technology in New Jersey, and the conclusions rendered in this report are of particular interest, and are quoted herein in full. In this analysis in order to simulate the effect of the compressive restraint provided in the three dimensional prototype, a homogeneous strut of xylonite was left

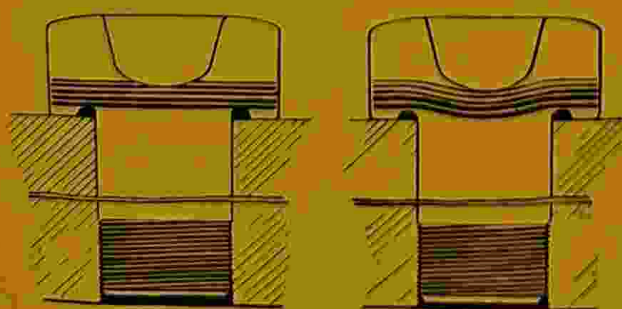


Figure 1—The sketches of the bolts show how the deformation of the head locks the bolt.

at the upper face of the specimens representing the sections of the Place head.

"1. It would appear that the value of frictional resistance of the Place head on its seat would be maintained over a larger arc of twist (in unscrewing) than could be expected with the standard head for the following reasons:

- a. The total elastic deformation appears greater in the Place head.
- b. In the standard head, the small region of stress concentration may result in the development of local spots stressed beyond the elastic limit with consequent permanent deformation.

"In our opinion, therefore, the Place head will be capable of resisting loosening action in use to a greater degree than the standard head.

"2. Since apparently the stresses are distributed through a larger volume and with lower local concentrations in the Place head, it is our opinion that the Place head would prove more suitable for impact loading and withstand fatigue to a greater degree than the standard head.

"It must be noted that these are qualitative opinions and no predictions as to the quantitative degree of superiority can be made."

A month or two later in 1941, a comparative vibration test was carried out between conventional bolts and Place type bolts in order to determine to what degree the Place head construction improved the resistance to loosening from vibration and to study its ability to withstand fatigue stresses. The procedure followed in this test may be of interest to the reader.

Procedure—Standard bolts 7/16 in.-20 x 1 1/4 in. and Place bolts of the same size formed from SAE 1035 stock, heat treated, were furnished for these tests. Both types of bolts were checked for hardness, breaking load, strength, and torque wrench measurements of tightening torque and loosening torque.

For the vibration testing of these bolts a special apparatus was assembled consisting of a heavy steel ring rigidly bolted to one end of the vibration motor, and upon this ring was mounted a 1/4 in. steel plate attached by the bolts under test. The setup was vibrated at its resonant frequency.

The frequency of vibration was determined by the use of a calibrated continuously variable oscillator as the source of the alternating power. This initiated vibratory motion and varied the frequency until resonance was noted. The alternating forces to vibrate the steel plate were produced by an electromagnetic vibration motor. In this manner fairly

high stresses were developed under the heads of the test bolts with relatively little bending. The motion at the center of the 1/4 in. steel plate was 0.050 in. and the motion of the plate at the head of the bolt was less than 0.001 in. The test bolts, lubricated with oil and fitted with a hardened flat washer under the head, were assembled through body holes in the steel plate into tapped holes in the heavy steel ring. Each bolt was tightened to a torque of 75 ft lb with a calibrated 100 ft lb capacity torque wrench, released to measure the initial loosening torque and again tightened to 75 ft lb. The amplitude of motion of the steel plate was measured with an accuracy of 0.001 in. by means of an accurate gage and the plate was then vibrated at a constant amplitude of 0.050 in.

Results—After 5,800,000 cycles of vibration in Test A, and after 10,000,000 cycles in Test B, the Place bolts showed no reduction of loosening torque. As a matter of fact, in each case the loosening torque after vibration was slightly greater than the initial loosening torque. However, the loosening torque of the standard bolts showed an appreciable loss after 5,800,000 cycles and a loss of 26 ft lb after 10,000,000 cycles. The overall length of the bolts was carefully measured both before and after testing to determine whether any permanent elongation took place as the result of the tightening torque or the vibration stresses. The amount of permanent elongation was very small and no correlation could be noted between the amount of permanent elongation and the loosening torque for either type of bolt.

Conclusions—In bolt assemblies where vibration stresses cause a reduction of the loosening torque of standard bolts there is no reduction in the loosening

CONVENTIONAL

PLACE BOLT HEAD

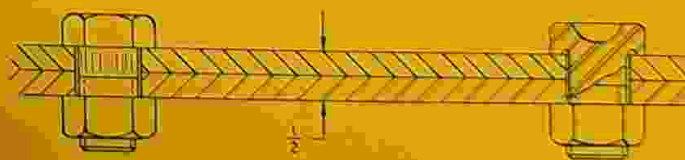


Figure 2—The bolt on the left is of the conventional type and at a stress of 100,000 psi has an elastic elongation in the shank of 0.0016 in. The bolt on the right, which is of the Place type and at the same stress, has the same elongation in the shank plus an elongation in the head of 0.005 in. for a total elastic elongation of 0.0066 in. or 300 per cent more than the conventional.

ing torque of these bolts. Static tensile tests prove that the Place head and the standard head have equivalent strength and that under tensile load, the fracture always occurs in the threaded section of the bolt.

In considering the foregoing conclusions resulting from the tests, it is interesting to consider the practical application of a Place type bolt head. A typical example is a short bolt used in conjunction with a nut as shown in Figure 2. Shape and proportions of the diaphragm section enable the bolt head to flex elastically in the direction of the bolt axis an amount which is roughly in direct proportion to the axial stress of the bolt. The ratio of this additional elastic bolt elongation to bolt stress is of the order of 0.005 for 100,000 psi stress in all bolt sizes. Thus, the added elasticity gained by employing Place bolts becomes a fixed percentage for each bolt length under stress, regardless of diameter.

If through wear, embedding or corrosion of bearing faces of either head or nut, the members of the assembly are reduced in thickness by 0.001 in. there is in the case of the conventional bolt only a small fraction of the original bolt tension left. The nut is apt to become loose, or in the case of a dynamically loaded fastener, fatigue failure may follow. The other bolt, in the example shown, however, under the same conditions will have lost less than 1/6 of its original tension and neither fatigue nor loosening of nut will result.

The curve of Figure 3 shows the advantages to be gained by substituting this type head for conven-

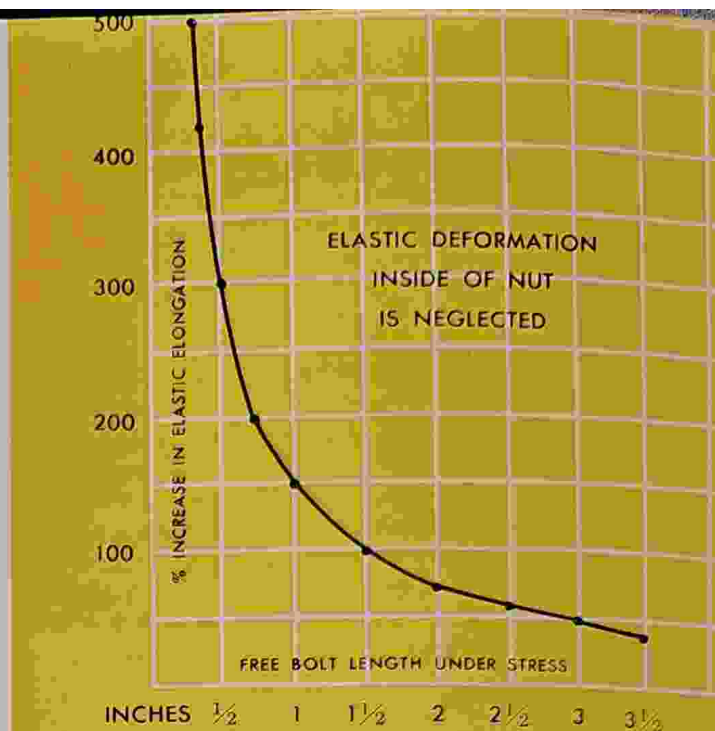


Figure 3—The curve indicates how much elastic elongation is gained by employing a Place bolt of a certain length instead of a conventional bolt of the same length. This added amount varies with the stress but is always a fixed percentage of the elastic elongation of the stressed shank portion.

tional heads in both bolts and capscrews, and especially so in the case of short length parts.

It should always be borne in mind that high spots which may occur on the underside of bolt or capscrew heads, or on the underside of nuts may not be fully compressed at time of shop assembly, but later due to working or abrasive action on the road, these high spots may completely wear off. Under such conditions bolts or capscrews, constructed so that the heads give added elastic elongation, offer added protection against complete loss of initial bolt tension under such contingencies.

In sizes 1/2 in. and upwards, this type bolt is extensively used for securing crankshaft main bearing caps, generally in diameter from 1/2 to 5/8 in. inclusive, but in certain of the larger engines these bolts are being used for the bearing cap application in 7/8 in. diameter sizes, and on the same engines these bolts of 3/4 in. diameter sizes are used for securing connecting rod caps. A very general use of this product is being made for securing flywheels to crankshafts in sizes from 7/16 to 9/16 in. inclusive, and generally speaking this application is an example of the advantage of using this type

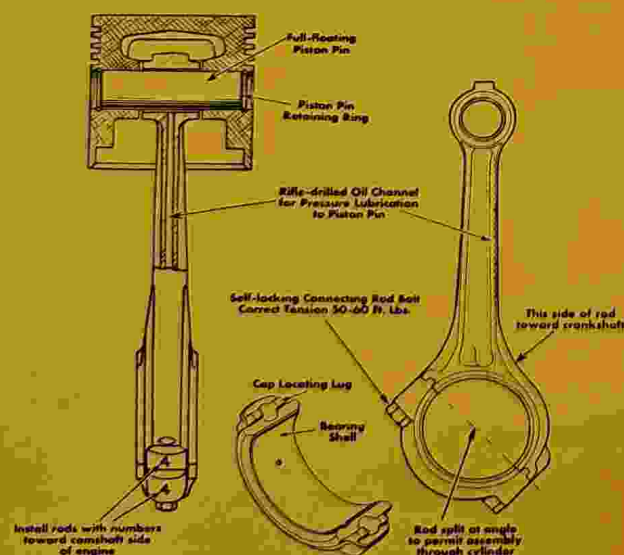


Figure 4—This sketch of a manufacturer's connecting rod and piston assembly shows one use for the self-locking bolt.

head, not only on account of its locking qualities but because most bolts used in this application are of short free length.

During the war period, large quantities of 1 inch diameter bolts made to this design were used in tank suspensions. Bolts of smaller diameter, but to the same design, were used for securing driving sprockets and other parts of tank assemblies.

It is also interesting to note that on account of the flexing qualities developed in the head design, very large quantities of 5/16 in.-24 Place bolts were used in conjunction with slotted nuts and wiring for securing collector ring flanges in aircraft construction. In this particular application, the elastic properties of the head compensated for expansion and contraction through a wide range of operating temperatures.

Reference to the standards sheets of all major automotive manufacturers in which Place head bolts are dimensionally specified will show that the head is actually slightly shallower than the standard head. This feature, of course, is advantageous when a designer is limited by close clearances. There is also a considerable saving in weight when the Place heads are substituted for conventional heads, as reference to Table I indicates.

While the weight saving feature may not be of so much importance in automotive construction, it would appear to be of definite importance in aircraft construction as shown in a paper read before the Society of Automotive Engineers in April, 1943, and later confirmed by another aircraft authority in the SAE Journal of May of the same year. These papers indicated that each pound of weight saved on a commercial plane increases its earning capacity by \$100.00 a year.

There are also many other applications where the Place type head not only performs as a locking unit, but where it also acts as a preventive to oil leakage. Quite obviously by virtue of its unit construction, it may be used without hesitation internally in gear cases or transmission without the fear of fracture and resultant gear stripping which might take place if a capscrew with a separate locking medium were used. This particular type of application can show very substantial savings in designs where it has been the practice of designers to countersink or recess the capscrew seating face for the entry of the locking medium, when such operations have been undertaken purely for the prevention, in the event of breakage of the locking medium, of the fractured portion falling into the gears.



Figure 5—The Place bolt is here used in a fly-wheel-to-crankshaft application.

TABLE I

**WEIGHT SAVINGS
PLACE BOLTS VS STANDARD BOLTS
AUTOMOTIVE STANDARDS**

Size	Th'ds Per Inch	Nominal Head Width	Nominal Head Thickness	Max. Savings In Lbs. Per 1,000 Pieces As Compared With Standard Bolts	Per Cent Reduction In Bolt Head Alone
1/4	20	3/8	3/8	1.00	17
	28			1.00	17
5/16	18	1/2	5/8	2.40	21
	24			2.20	19
3/8	16	3/4	7/8	4.80	25
	24			4.60	24
1/2	14	1 1/4	1 1/8	7.50	28
	20			6.60	25
5/8	13	1 3/4	1 1/2	10.20	24
	20			9.40	22
3/4	12	2	1 3/4	16.00	27
	18			14.70	24
7/8	11	2 1/4	2	20.00	19
	18			17.50	17
1	10	2 3/4	2 1/4	40.00	24
	16			31.50	19
1 1/8	9	3	2 3/4	51.00	21
	14			47.00	19
1 1/4	8	3 1/4	3	70.00	20
	14			65.00	18

AIRCRAFT STANDARDS

Size	Th'ds Per Inch	Nominal Head Width	Nominal Head Thickness	Max. Savings In Lbs. Per 1,000 Pieces As Compared With Standard Bolts	Per Cent Reduction In Bolt Head Alone
5/16	12	1/2	5/8	6	16
3/8	18	3/4	7/8	1.5	22
1/2	24	1 1/4	1 1/8	2.4	22
5/8	24	1 3/4	1 1/2	3.8	22
3/4	20	2	1 3/4	6.0	25
7/8	10	2 1/4	2	8.7	23
1	18	2 3/4	2 1/4	12.0	21
1 1/8	10	3	2 3/4	16.0	21
1 1/4	16	3 1/4	3	26.0	23
1 1/2	14	3 3/4	3 1/4	44.0	25
1 3/4	14	4	3 1/2	58.0	21

